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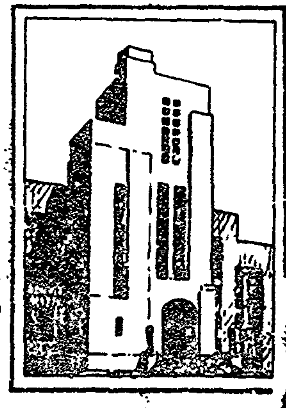

DEPARTMENT OF THE NAVY
DAVID TAYLOR MODEL BASIN

MECHANICS

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APPLIED MATHEMATICS

EXPERIMENTAL STRESS ANALYSIS OF A SOCKETED CONNECTION IN BENDING

by

Louis A. Becker

STRUCTURAL MECHANICS LABORATORY

RESEARCH AND DEVELOPMENT REPORT

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ABSTRACT

This report presents a semiempirical curve for use in designing socketed connections subjected to pure bending. This type of connection is frequently used in rudder and control surface connections on surface ships and submarines. The curve is for use only within a limited range of parameters considered of immediate interest to the Bureau of Ships. These include the depth of penetration of the shaft into the socket, the thickness of the socket wall, the presence of relief in the socket wall, and the type of attachment of the stock to the socket. The design curve was obtained by measuring strains at discrete locations on the socketed connection loaded in pure bending. The tests were repeated as the various parameters were changed. The resulting design curve enables the user to predict the maximum flexural stress in the socketed connection with a stock penetration of one stock diameter or more.

INTRODUCTION

One of the problems facing the ship designer is the socketed connection, a type used in ship construction. Two examples of its use are in the connection of rudders to their stocks and in the connection of diving planes to their stocks. Most of these designs require the insertion of a tapered shaft into a matching socket. The shaft is secured either by a nut on the end of the shaft or by a tapered key driven through the assembly.

At the present time, most designs call for a penetration of 2 or 2 1/2 stock diameters. In order to develop the surface friction of the joint, the two pieces must be fitted so that they have a metal-to-metal contact of at least 80 percent. This necessitates machining both the tapered end of the stock and the inside of the socket to a very smooth finish (16 rms or better) and usually requires hand finishing during assembly. Considering the sizes involved on some of the newer ships, it is obvious that such an assembly is both large and very costly. The problem was aggravated by the hull shape and the use of large single propellers with shafts of large diameter developed for USS ALBACORE and subsequent submarines. These hulls, which narrow toward a point at the stern, do not have sufficient space for the penetrations desired. Thus, considerable interest has been shown in improving the socketed connection design, particularly along the lines of making the assembly smaller and lighter.

Because of the need for a design procedure for this type of structure, the David Taylor Model Basin was requested[1] to determine the governing design criteria for this type of joint under pure bending. Specifically, the investigation was to determine:

a. If the entire joint assembly should be treated as two interacting beams or as a single monolithic structure.

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[1]References are listed on page 23

b. If there is any appreciable difference in the strength of the joint when the middle third of the tapered fit is relieved.

c. The variation in the strength of the joint as the length of the taper engagement is varied from 1/2 to 3 stock diameters.

d. The difference in cost between a rudder nut and a taper key connection.

Accordingly, tests were conducted at the Model Basin in 1960 and 1961 to investigate the above requirements. This report presents the results of an investigation conducted on model scale and includes the effects of stock penetration, yoke* wall thickness, relief, and type of stock attachment (rudder nut or tapered key). Within the limits of the investigation, some basic conclusions about the design methods are drawn.

DESCRIPTION OF MODELS

The experimental program employed scaled models of typical socketed connections. The models were designed so that it would be possible to investigate the effects of different stock penetrations, types of connections, yoke wall thicknesses, and relief. Since the proposed new stern arrangement of USS ALBACORE[2] (AGSS 569) was of particular interest, it was used as the basic structure for scaling the models.

All the models had the same general configuration and consisted of two shafts or stocks made of 4140 steel. Material properties are given in Table 1. In general, the models were scaled to one-fourth the size proposed for ALBACORE. The model stock diameters were 4 1/2 in. Each stock had a 2 in./ft of diameter taper on one end, and the two stocks were joined by a yoke as shown in Figures 1 and 2. The tapered surfaces were ground to provide at least 80 percent contact of the yoke with the stocks. The stocks were attached to the yoke either by a tapered key or by a rudder nut, as shown in Figure 3. One of the yoke wall thicknesses was scaled from that proposed for ALBACORE and a second yoke scaled to 0.6 its thickness was made to determine the effects of reduced yoke wall thickness. Duplicate models were made to determine the effect of adding a relief to the middle third of the yoke except at 1 1/4-diameter penetration.

TABLE 1

Material Properties

Piece	Tensile Yield psi	Tensile Ultimate psi	Compressive Yield psi
Yoke 1	42,500	93,100	43,100
Yoke 2	41,900	93,500	45,400
Yoke 3	42,900	94,000	44,200
Yoke 4	-	-	48,500
Stock 1	82,900	111,200	95,500
Stock 2	80,600	108,800	90,600

*Throughout this report, the piece with the internal tapered sockets will be referred to as the yoke.

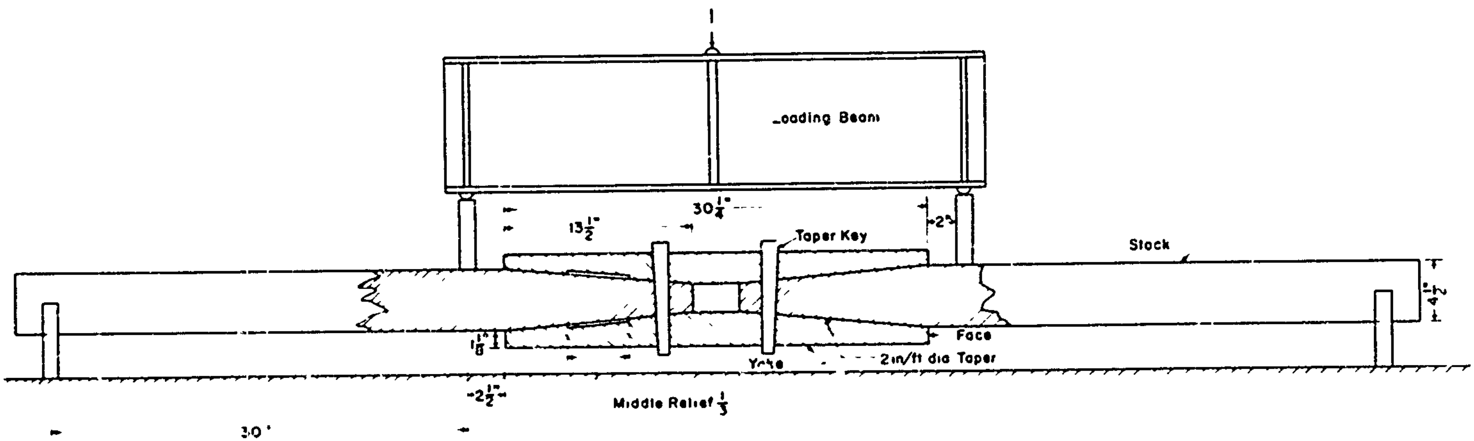


Figure 1 – Model Used for a Depth of Penetration of Three Diameters

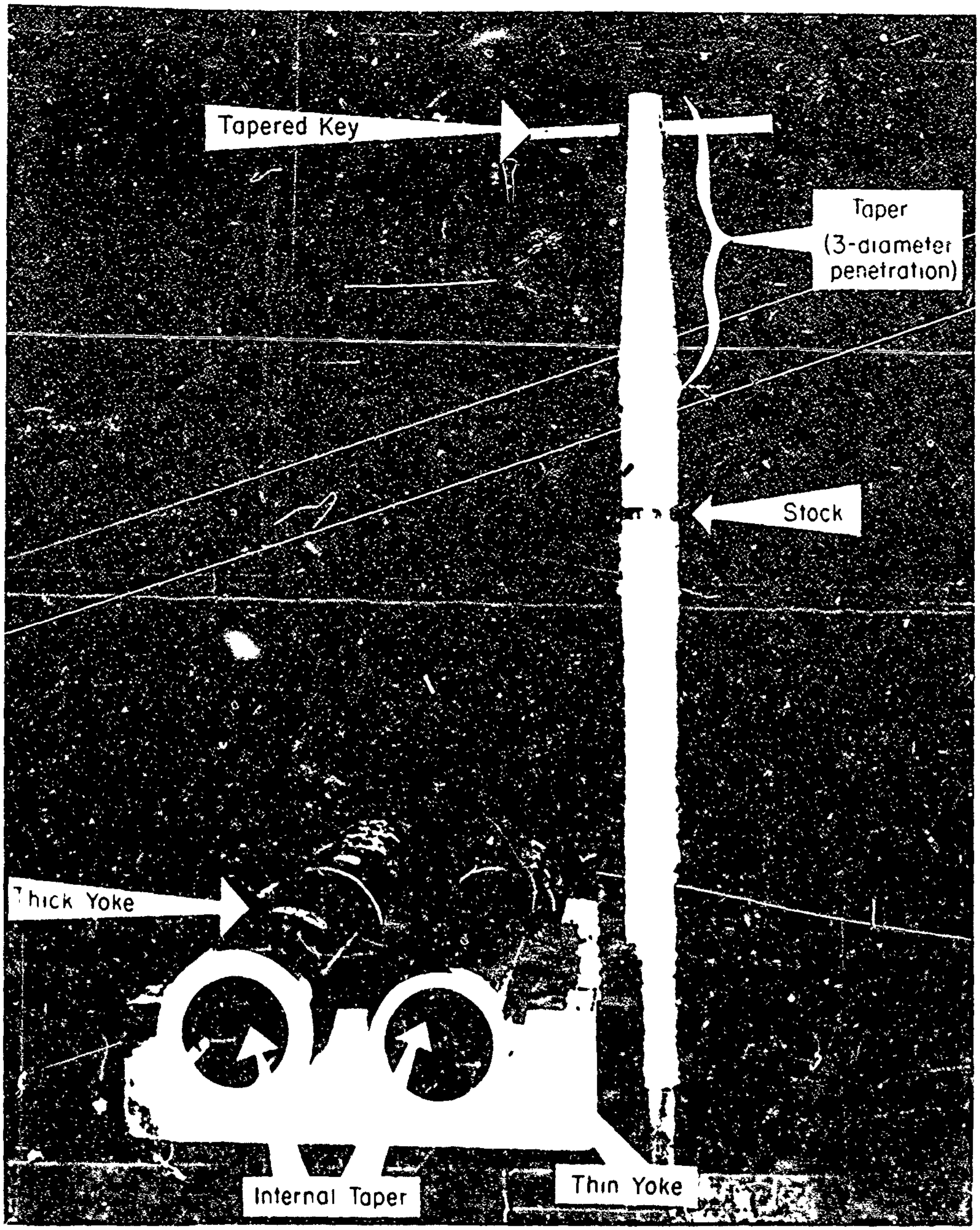


Figure 2 – Model Components

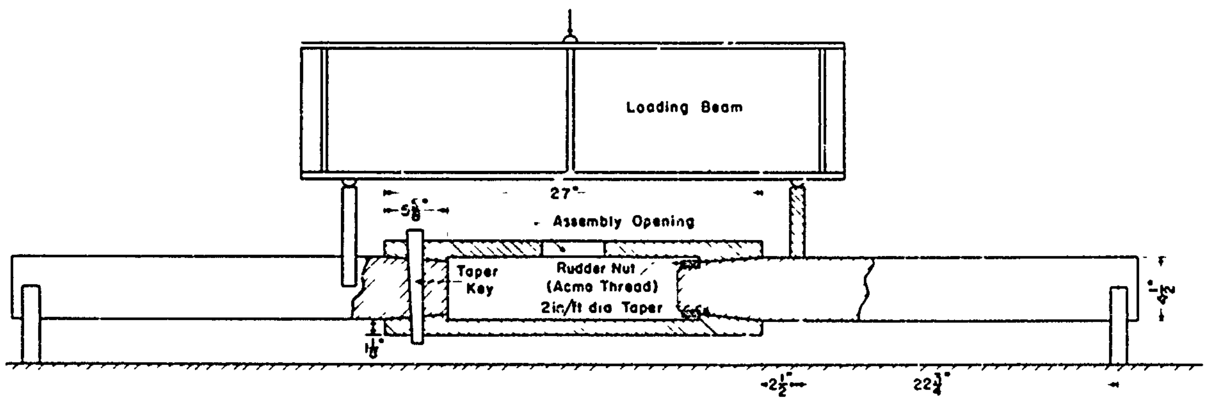
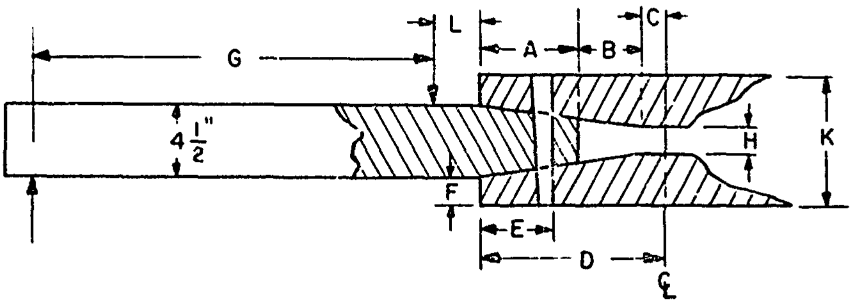


Figure 3 – Model Used to Compare Rudder Nut and Tapered Key

Pertinent dimensions for all models are given in Table 2. To economize on model construction costs, two basic stocks and four yokes were systematically modified during the course of testing to provide the required variations in parameters for the 18 models listed in Table 2.

TABLE 2

Principal Test Dimensions



Model Number	A in.	B in.	C in.	D in.	E in.	F in.	G in.	H in.	K in.	L in.
1 [illegible]	13 1/2 or 3 Dia	1/2	1 1/4	15 1/4	12 1/2	1 1/8	30	2 1/8	6 3/4	2 1/2
[illegible]	13 1/2 or 3 Dia	1/2	1 1/4	15 1/4	12 1/2	11/16	30	2 1/8	5 7/8	2 1/2
5 6	4 1/2 or 1 Dia	9 1/2 3/4	1 1/4 1	15 1/4 6 1/4	3 1/2	1 1/8	30	2 1/8 3 5/8	6 3/4	2 1/2
7 8	4 1/2 or 1 Dia	9 1/2 3/4	1 1/4 1	15 1/4 6 1/4	3 1/2	11/16	30	2 1/8 3 5/8	5 7/8	2 1/2
9 10	2 1/4 or 1/2 Dia	11 3/4	1 1/4	15 1/4	1 5/8	1 1/8	30	2 1/8	6 3/4	2 1/2
11 12	2 1/4 or 1/2 Dia	11 3/4	1 1/4	15 1/4	1 5/8	11/16	30	2 1/8	5 7/8	2 1/2
13 14	5 5/8 or 1 1/4 Dia	0	7 7/8	13 1/2	2 3/4	1 1/8	22 3/4	4 1/2	6 3/4	2 1/2
15 16	5 5/8 or 1 1/4 Dia	0	7 7/8	13 1/2	-	1 1/8	22 3/4	4 1/2	6 3/4	2 1/2
17 18	5 5/8 or 1 1/4 Dia	0	7 7/8	13 1/2	2 3/4 -	1 1/8	22 3/4	4 1/2	6 3/4	2 1/2

INSTRUMENTATION AND TEST PROCEDURE

Prior to testing, each model was instrumented with strain gages as shown in Figure 4. Two basic groups of gages were used. The first group was used to compare longitudinal bending strains in the stock and yoke with theory and to check the symmetry of loading. The second group was used to measure the face strains in order to determine bearing stresses.

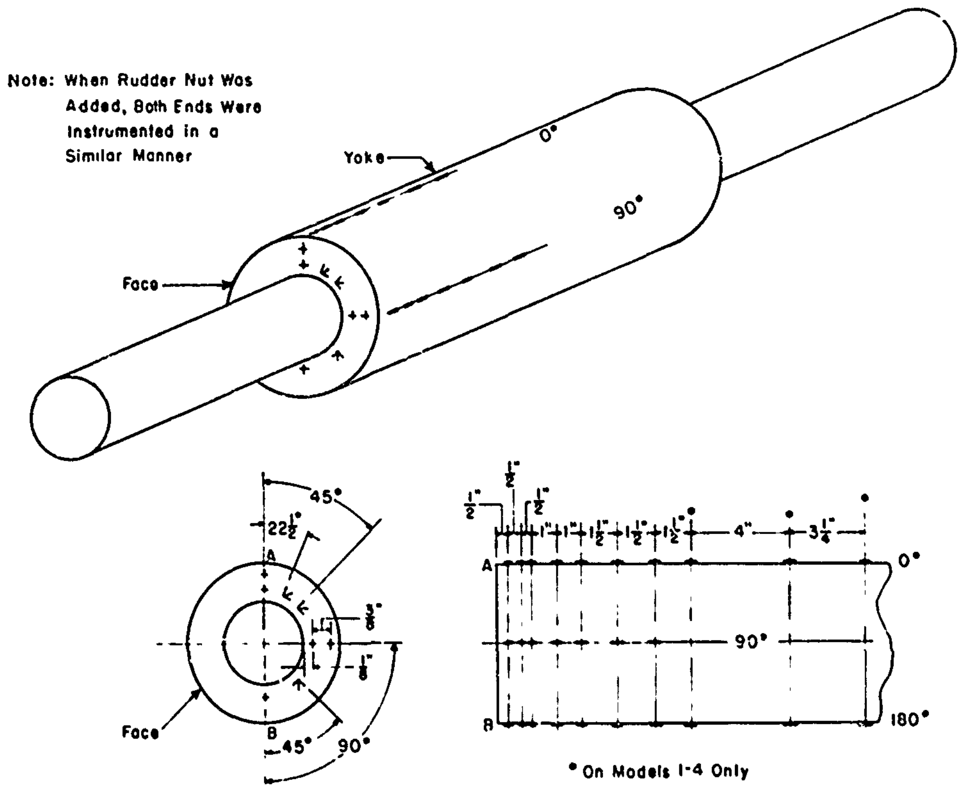


Figure 4 -- Strain Gage Locations for Socketed Joint Connections

Each model was tested in the 600,000-lb testing machine. The basic test setup is shown in Figure 5. The extreme ends of the model were supported on knife edges. The model was subjected to pure bending across the test section by applying the load through the loading beam. The loading beam was used to keep the load symmetric. No torque or shear was applied at the yoke.

The test of each model consisted of loading the model in increments of 1000 or 2000 lb. All strain gages were read at each increment of load. This procedure was continued until a maximum strain of 1000 μin./in. was observed. The tests were stopped at this point since no yielding could be permitted because the various pieces were to be used again.

Figure 5 — Test Setup

This procedure was followed on all tests listed in Table 3.

Up to this point all testing was static. However, there is considerable vibration of th stocks in a ship due to hull frequency, blade frequency, shaft speed, etc. The amplitude an frequency of this vibration varies considerably from ship to ship, and no given vibratory

TABLE 3

Test Schedule

Test Number*	Model Number	Penetration in stock diameters	Yoke	Relief	Attachment
3LTR	2	3	Number 1 thick	yes	Taper Key
3STR	4	3	Number 1 thin	yes	Taper Key
3LT	1	3	Number 2 thick	no	Taper Key
3ST	3	3	Number 3 thin	no	Taper Key
1.25LT	13	1 1/4	Number 2 thick	no	Taper Key
1.25LN	15	1 1/4	Number 2 thick	no	Rudder Nut
1.25(90)LT	17	1 1/4	Number 2 thick	no	Taper Key
1.25(90)LN	18	1 1/4	Number 2 thick	no	Rudder Nut
1.25(A)LT	14	1 1/4	Number 2 thick	no	Taper Key
1.25(A)LN	16	1 1/4	Number 2 thick	no	Rudder Nut
1LTR	6	1	Number 4 thick	yes	Taper Key
1STR	8	1	Number 4 thin	yes	Taper Key
1LT	5	1	Number 2 thick	no	Taper Key
1ST	7	1	Number 3 thin	no	Taper Key
0.5LT	9	1/2	Number 2 thick	no	Taper Key
0.5ST	11	1/2	Number 2 thin	no	Taper Key
0.5LTR	10	1/2	Number 2 thick	yes	Taper Key
0.5STR	12	1/2	Number 2 thin	yes	Taper Key

*The meaning of the test number is as follows

1.25 – Number of stock diameters of penetration of stocks into yoke.
(90) – Special consideration – (A) indicates tests run after vibration of assembly, (90) indicates taper key turned vertical instead of on neutral axis.
L – Thickness of yoke – L is thick, S is thin.
T – Type of attachment – T is taper key, N is rudder nut.
R – Relief.

loading can be considered as representative for all ships. However, it was felt that vibration should not be neglected. Accordingly, a random sinusoidal vibration with a maximum alternating force of ± 1000 lb and a maximum frequency of 25 cps was applied to the center of the model by means of a Lazan[3] oscillator. This vibratory load was applied for 8 hr to a thick yoke model having a depth of penetration of 1 1/4 diameters. The model used had a tapered key in one stock and a rudder nut in the other stock. Bending loads were applied to the model before and after vibration and the strains were compared.

TEST RESULTS

The bending strains measured on the extreme fiber of the yoke during these tests are converted to stresses directly from the following equation:

$$\sigma = E\epsilon \quad [1]$$

where σ is stress in psi,

E is modulus of elasticity in psi, and

ϵ is strain in in/in.

The resulting stresses for the models with tapered key are plotted in Figures 6 through 12. Figure 13 compares the stresses resulting from a tapered key connection with those resulting from a rudder nut connection. All these strains were taken at an applied moment of 75,000 in-lb. This was the highest moment that could be safely applied to all models without causing yielding. In each of these figures the theoretical stresses, computed by assuming the structure to be monolithic, were also plotted. These were computed from the equation:

$$\sigma = \frac{Mc}{I} \quad [2]$$

where σ is the stress in psi,

M is applied moment in in-lb,

I is moment of inertia in^4, and

c is the distance from the neutral axis to the extreme fiber in inches

An examination of Figures 6 through 13 indicates differences between the theoretical and experimental bending stresses. The differences become quite large as the length of penetration is reduced. The results shown in these figures are summarized in Table 4 which indicates the maximum experimentally determined stress, the corresponding theoretical stress computed by Equation [2], and the correction factor K, which is the ratio of these two stresses. The variation of K with penetration is plotted in Figure 14. It must be noted that all the results are based on Equation [1]. This is not strictly correct when the depth of penetration is very small or when the yoke is very thin. Under these conditions there may also be some high circumferential stresses. This is evident by the sign reversal in the bottom fiber of the small depth of penetration models. No allowance was made for this since it was felt that the stresses were small compared to the maximum bending and that their effects were accounted for in the correction factor. This is one reason for limiting the use of Figure 14 to one-diameter penetration or more.

Other types of data also collected during the tests included face stresses (Tables 5 and 6) and bending results before and after vibration (Figure 15). One result of the vibration tests was that the rudder nut was loose at the end of the 8 hr of vibration.

(Text continued on page 18.)

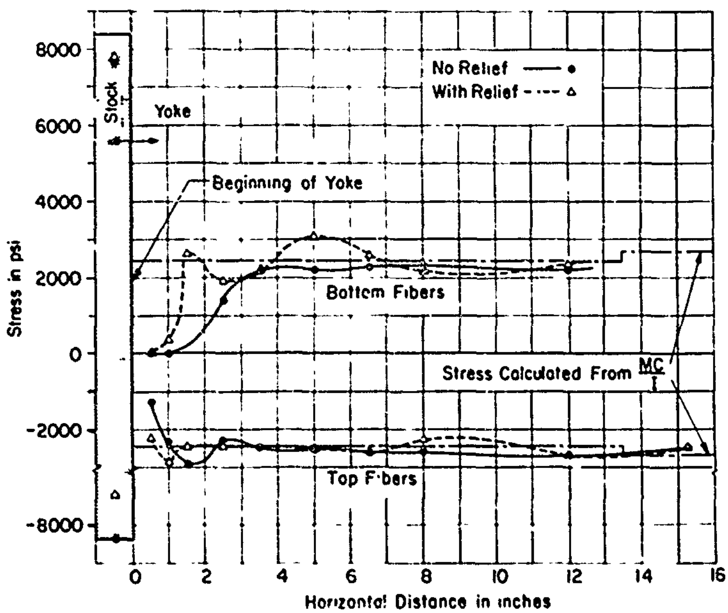


Figure 6 – Bending Stress Distribution for Three-Diameter Penetration, Thick-Wall Yoke, 75,000 In-Lb

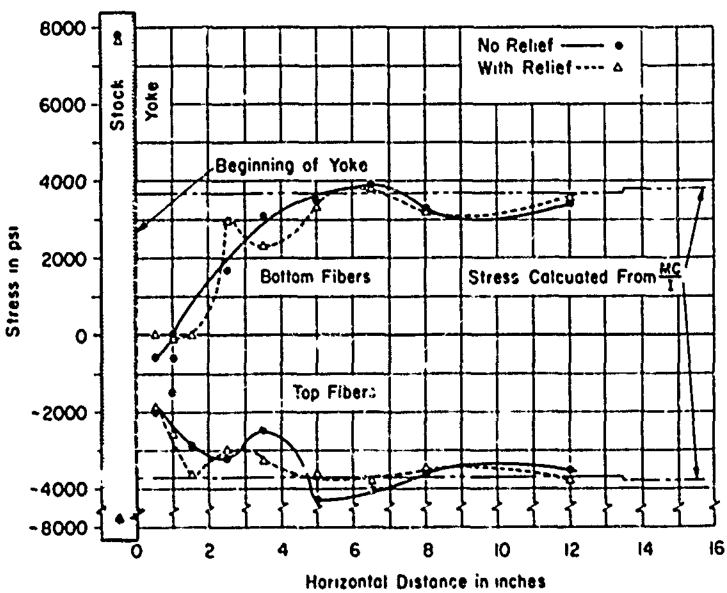


Figure 7 – Bending Stress Distribution for Three-Diameter Penetration, Thin-Wall Yoke, 75,000 In-Lb

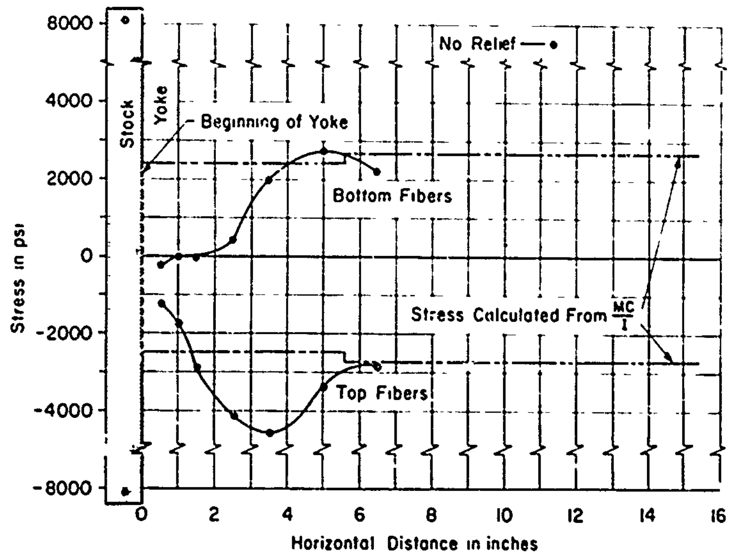


Figure 8 – Bending Stress Distribution for 1 1/4-Diameter Penetration, Thick-Wall Yoke, 75,000 In-Lb

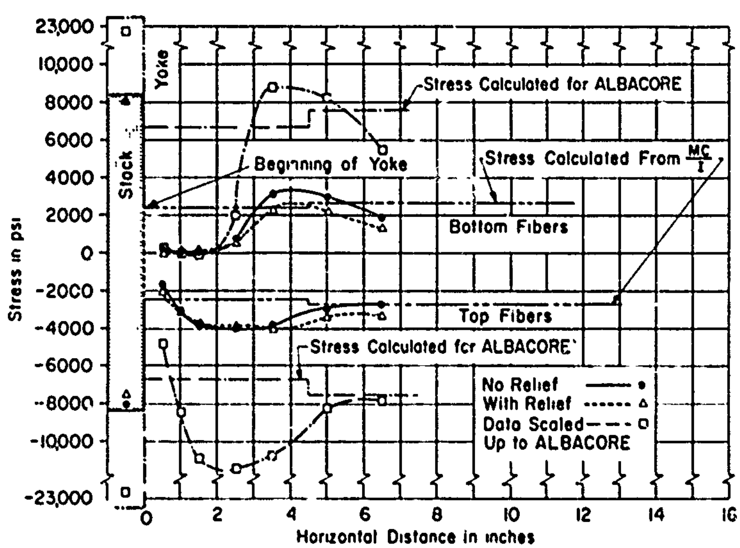


Figure 9 – Bending Stress Distribution for One-Diameter Penetration, Thick-Wall Yoke, 75,000 In-Lb

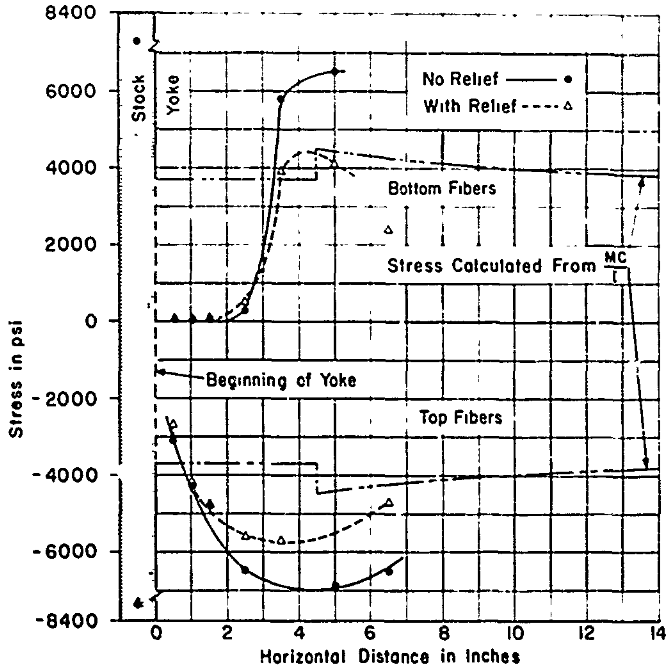


Figure 10 – Bending Stress Distribution for One-Diameter Penetration, Thin-Wall Yoke, 75,000 In-Lb

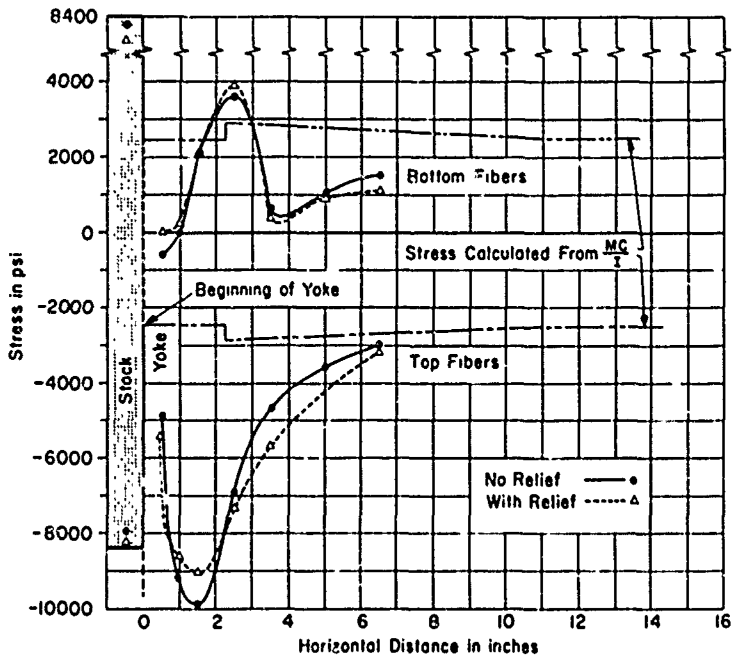


Figure 11 – Bending Stress Distribution for 1/2-Diameter Penetration, Thick-Wall Yoke, 75,000 In-Lb

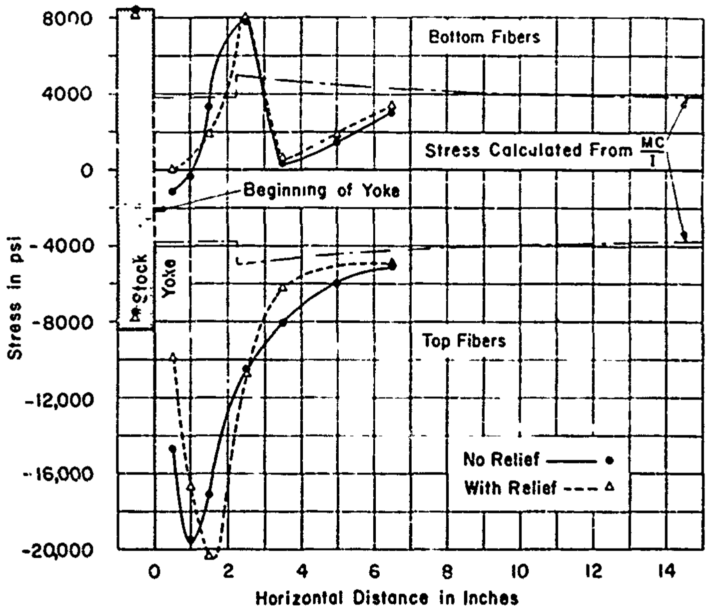


Figure 12 – Bending Stress Distribution for 1/2-Diameter Penetration, Thin-Wall Yoke, 75,000 In-Lb

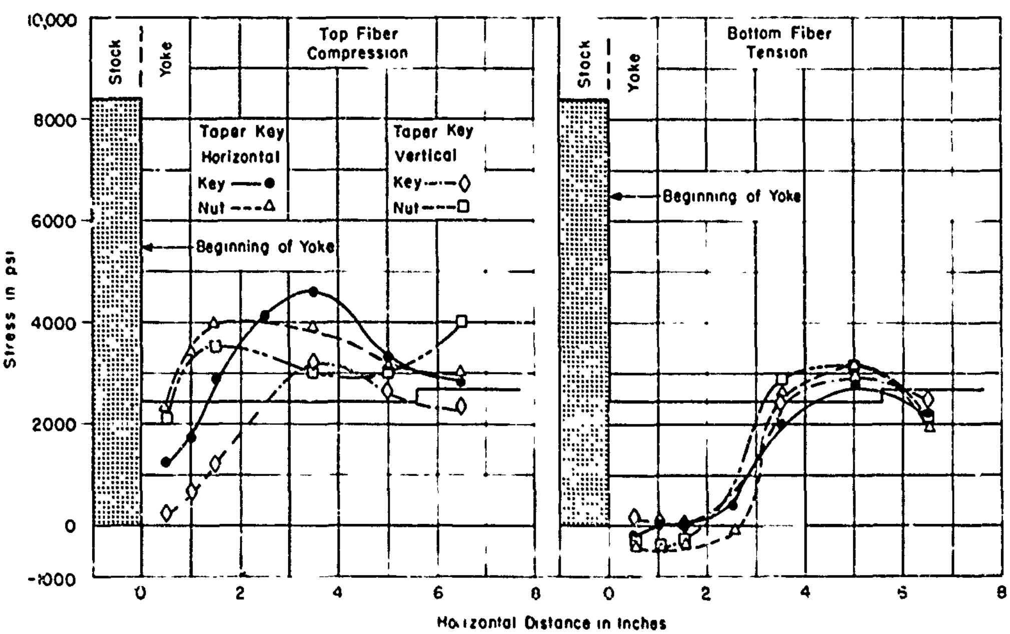


Figure 13 – Bending Stress Distribution for 1 1/4-Diameter Penetration, Thick-Wall Yoke, 75,000 In-Lb

TABLE 4

Bending Stresses

Test Number	Maximum Experimental Stress σ_e psi	Corresponding Theoretical Stress σ_t psi	$K = \frac{\sigma_e}{\sigma_t}$
3LTR	3100	2400	1.29
3LTN	2900	2400	1.21
3STR	3800	3700	1.03
3STN	4400	3700	1.19
1LTR	4000	2400	1.66
1LTN	4100	2400	1.71
1STR	5700	3700	1.54
1STN	7200	3700	1.95
0.5LTR	9000	2400	3.74
0.5LTN	9900	2400	4.11
0.5STR	20400	3700	5.51
0.5STN	19550	3700	5.28
1.25LNN	4000	2400	1.67
1.25LTN	4600	2400	1.92
1.25(90)LNN	3500	2400	1.46
1.25(90)LTN	3200	2400	1.33
1.25(A)LNN	4130	2400	1.72
1.25(A)LTN	4890	2400	2.00

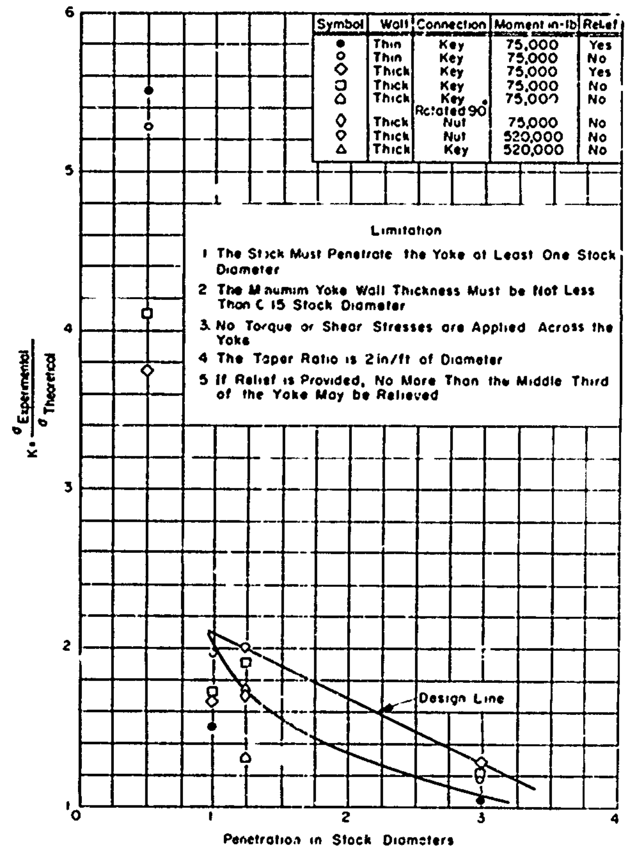
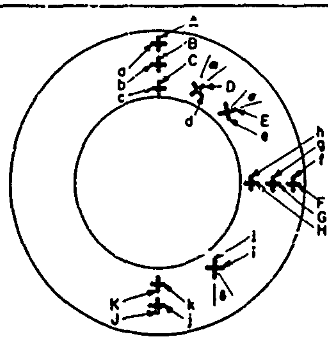


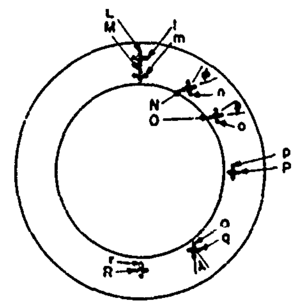
Figure 14 – "K" Factor versus Penetration

TABLE 5

Face Stresses at a Moment of 75,000 In-Lb

All stresses in psi; all angles in deg.





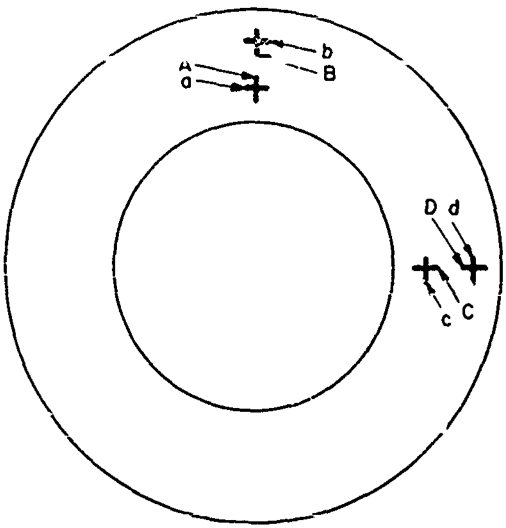
Location	3LTR	3LT	1LTR	1LT	0.5LTR	0.5LT	1.25L T or N	1.25(90) LT or LN	1.25(A) LT + LN
	Test								
A	- 500	- 500	-	0	- 2,000	- 1,000	-	-	-
a	+5000	+4000	-	+5000	+12,000	+12,000	-	-	-
B	-2000	-3000	- 3000	-4000	- 9,000	- 7,000	+ 250 -19,300	+1250 -	+ 6,200 + 500
b	+4000	+5000	+2000	+3000	+12,000	+13,000	+ 1,900 + 1,[illegible]00	+2300 -	+24,700 +14,000
C	-3000	-6000	-6000	-7000	-17,000	-26,000	- 4,500 -22,900	+1300 +1200	+ 3,350 + 1,650
c	+5000	+4000	+7000	+6000	+13,000	+11,000	+ 2,000 +17,500	+4300 +6550	+11,200 +15,500
D	-2500	-4500	-	-4000	-11,000	- 2,000	-	-	-
d	+5000	+4000	-	+6000	+ 9,000	+10,000	-	-	-
E	-	-3000	-5000	-3500	-12,000	-14,000	-	-	-
e	-	+4000	+4000	+4500	+ 9,000	+10,000	-	-	-
F	0	0	-1000	-	-	+ 500	-	-	-
f	+1000	+1000	-1000	-	-	+ 6,000	-	-	-
G	0	0	-1000	+1000	0	0	+ 250 -	-1900 +2200	- + 1,500
g	+2000	+2000	+3000	+4000	+ 7,300	+ 7,000	+ 3,900 -	+4600 + 660	- + 7,000
H	0	0	-4000	- 500	+ 1,000	+ 1,000	+ 1,200 + 800	+5700 +5700	+ 700 + 2,650
h	+4000	+4000	+5000	+6000	+14,000	+15,000	+ 4,750 + 5,650	+1050 + 600	+16,500 + 800
I	0	- 500	-	-	-	-	-	-	-
i	+ 500	+ 500	-	-	-	-	-	-	-
J	-	-	0	0	0	0	-	-	-
j	-	-	-1000	-1000	- 1,000	- 1,000	-	-	-
K	0	0	- 500	0	- 1,000	- 1,000	-	-	-
k	-1500	-1500	-3500	-2000	- 3,000	- 4,000	-	-	-
α	0	7	-	7	7	8	-	-	-
γ	-	8	[illegible]	2	10	10	-	-	-
δ	58	45	-	-	-	-	-	-	-

Location	3STR	3ST	1STR	1ST	0.5STR	0.5ST
	Test					
L	-1000	- 200	-	-	- 5,000	- 5,000
l	+7000	+7000	-	-	+22,000	+28,000
M	[illegible]	-3000	- 3,000	- 2,000	- 6,000	- 9,000
m	+4000	+6000	+12,000	+11,000	+15,000	+27,000
N	0	-2000	-	- 2,000	- 8,000	-21,000
n	+7000	+6000	-	+ 7,000	+16,000	+17,000
O	-1000	-3000	0	- 2,000	- 7,000	-18,000
o	+6000	+5000	+10,000	+ 7,000	+13,000	+19,000
P	-1000	-1000	- 3,000	- 1,000	-	- 3,000
p	+4000	+3000	+ 5,000	+ 5,000	-	+ 1,800
Q	0	-	-	+ 1,000	-	+ 1,000
q	- 100	-	-	- 500	-	- 1,000
R	- 200	0	0	0	+ 1,000	- 500
r	-2000	-2000	- 3,[illegible]	- 2,000	- 6,000	- 9,000
φ	3	4	-	- 17	1	5
θ	- 8	-12	-11	- 3	- 5	+ 1
λ	23	-	-	41	-	55

TABLE 6

Face Stresses Measured during Maximum Loading Test

The stresses shown in this table are determined elastically from measured strains. Therefore stresses above 50,000 psi are not valid.



Moment in lb-in.	Key End		Nut End	
	Gage*	Stresses, psi	Gage*	Stresses, psi
900,000	A	- 68,000	A	- 7,000
	a	+152,000	a	+157,000
	B	- 82,000	B	+ 70,000
	b	+136,000	b	+219,000
	C	- 40,000	C	- 3,000
	c	- 12,000	c	+ 71,000
	D	+ 3,000	D	-
	d	+ 34,000	d	-
540,000	A	0	A	+ 10,000
	a	+ 49,000	a	+ 47,500
	B	- 6,500	B	+ 17,000
	b	+ 38,000	b	+ 38,000
	C	- 6,600	C	+ 1,000
	c	- 2,000	c	+ 34,500
	D	+ 4,000	D	-
	d	+ 17,000	d	-

*Indicates gage location in diagram.

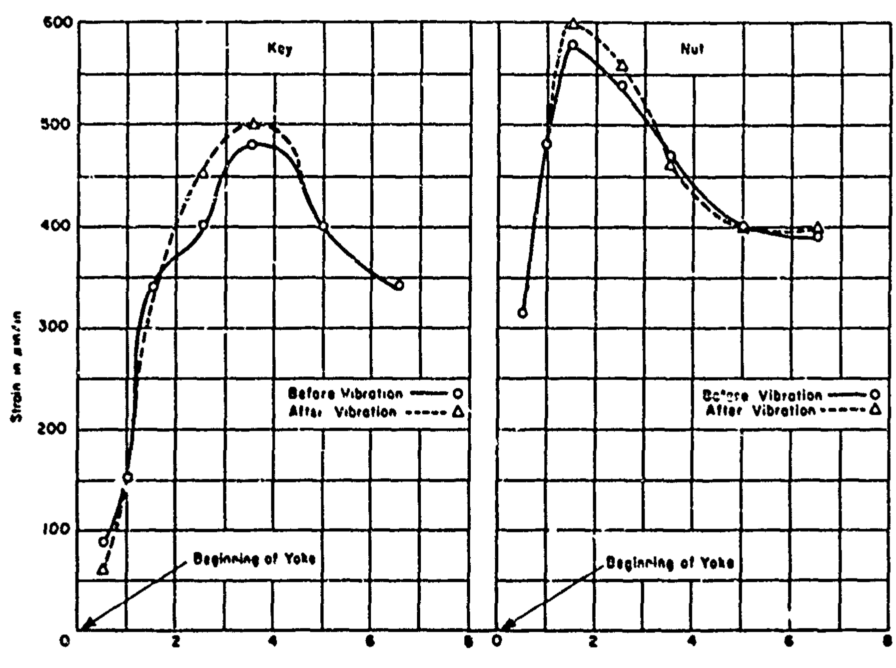


Figure 15 – Effects of Vibration on Top Fiber Strains of the Yoke

The final test was taken to failure. The results of this test are shown in Figures 16 through 21.

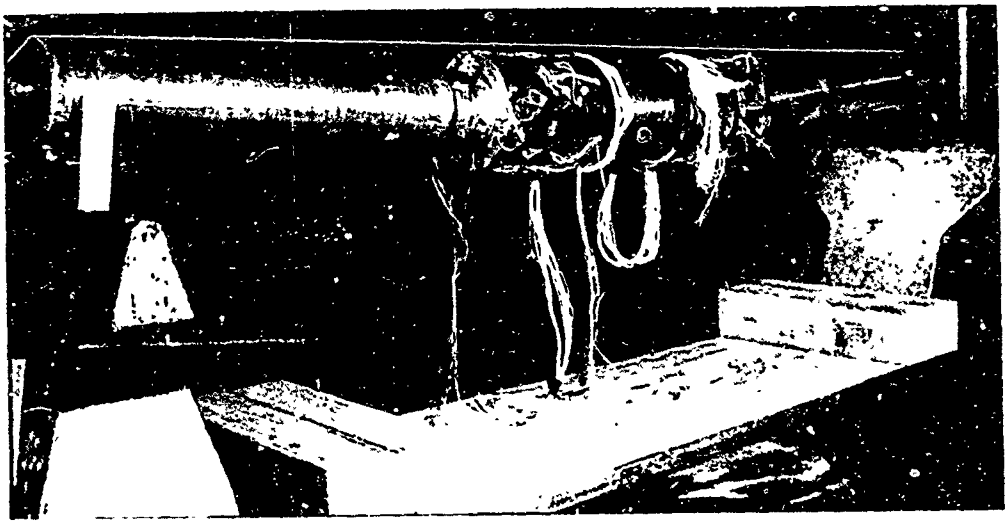


Figure 16 – Model after Maximum Applied Load

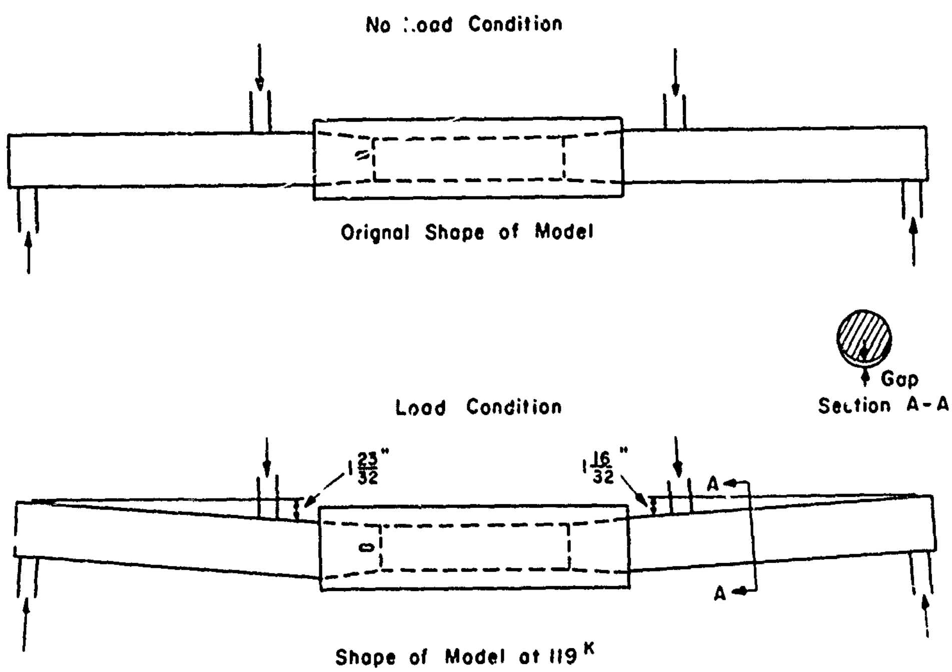


Figure 17 – Sketch of Damage to Model

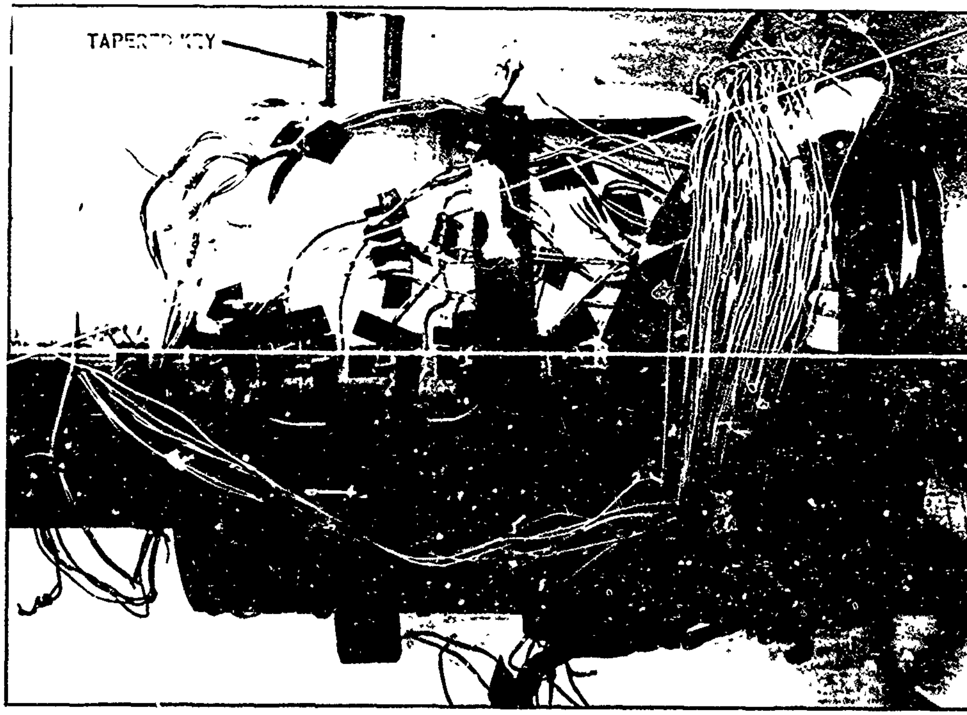
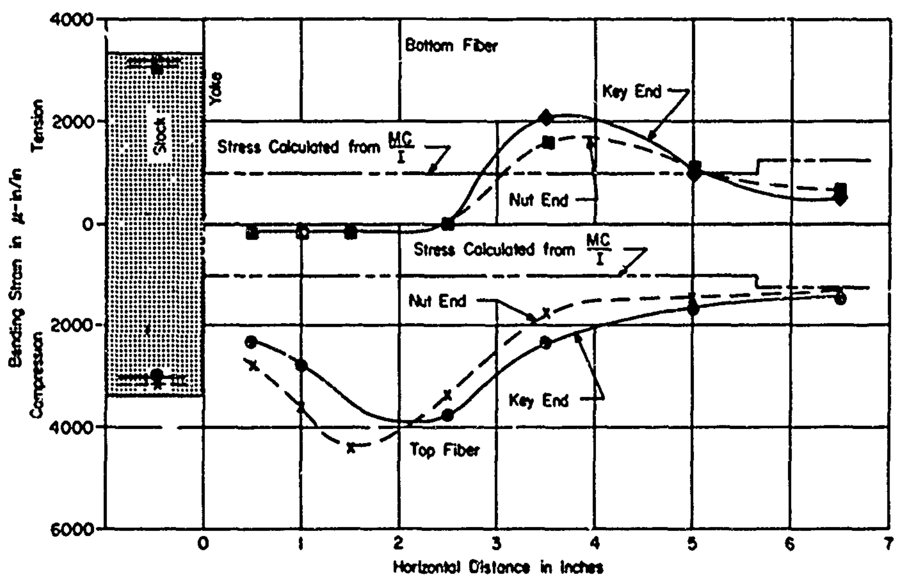


Figure 18 – Closeup of Damage to Tapered Key



ıgure 19 – Bending Strain Distribution for 1 1/4-Diameter Penetration, Thick Wall Yoke, at a Moment of 900,000 In-Lb

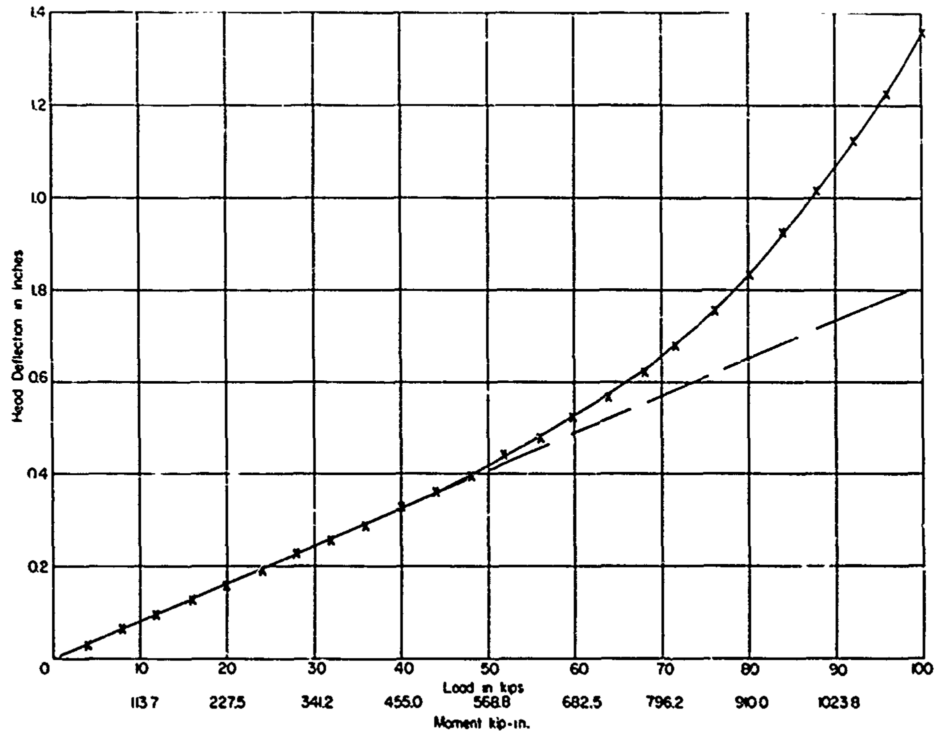


Figure 20 – Deflection of Head of Testing Machine versus Applied Load

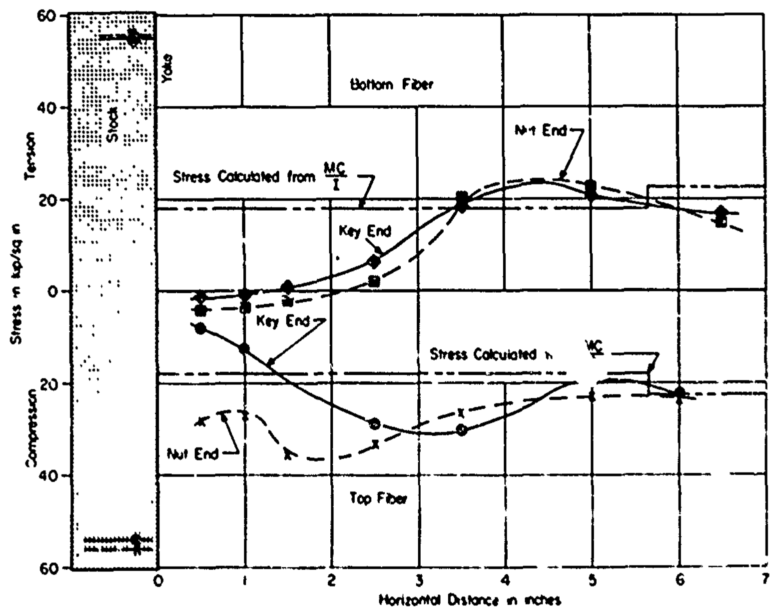


Figure 21 – Bending Stresses in 1 1/4-Diameter Penetration, Thick-Wall Yoke, at a Moment of 540,000 In-Lb

The models were disassembled after each test. To get some idea of the joint friction developed, one stock was removed from the assembly by loading the model in tension in the testing machine. The loads required to remove the stock from the yoke are tablulated below:

Penetration stock diameter	Load pounds
3	15,000 ± 1,000
1 1/4	not removed
1	7,500 ± 500
1/2	5,000 ± 500

DISCUSSION OF RESULTS

The pertinent results obtained from this program are summarized in Figure 14. Within the limits imposed by the experimental program, this curve can be used to predict the maximum bending stresses in a socketed connection. These limits are as follows:

1. The stock must penetrate the yoke no less than 1 stock diameter.
2. The minimum yoke wall thickness must be no less than 0.15 stock diameter.
3. No torques or shear stresses are applied across the yoke.
4. The taper ratio is 2 in/ft of diameter.
5. If relief is provided, no more than the middle third of the yoke may be relieved.

Although these limitations do not necessarily preclude the successful design of other connections, they do indicate that any great departure from them would require additional experimental verification.

Within these limitations, the bending stresses can be computed from the equations

$$\sigma = \left(2.5 - \frac{\psi}{2.5}\right)\frac{Mc}{I} \quad \text{when } 1 < \psi < 3.7 \qquad [3a]$$

or

$$\sigma = \frac{Mc}{I} \quad \text{when } \psi > 3.7 \qquad [3b]$$

where ψ is the stock penetration in stock diameters and σ, M, c, and I are defined in Equation [2]. Equations [3] are valid only in the elastic range.

As expected, the 1/2-diameter thin-wall yoke reached the 1000 μ in/in. limit at a lower load than any of the other models. This load was equivalent to an applied moment of 75,000 in-lb (35 percent of the design load for USS ALBACORE). Since this is the highest moment that could be applied to one model, it is the moment that should be used for comparison purposes. Therefore, the strains observed in each model at 75,000 in-lb are presented. Since the

model with the thick-wall yoke having a 1-diameter depth of penetration was scaled from the proposed ALBACORE design, it was also subjected to the proposed design moment, 210,000 in-lb, and the resulting strains were measured.

Several attempts were made to calculate the face stresses listed in Tables 5 and 6. However, this was abandoned as impractical because of the large number of variables present. Within the test limits, none of the bearing stresses are high enough to cause concern but they should be considered in any extreme design.

During the majority of these tests, the tapered key was on the neutral axis so that the extreme fiber strains would not be influenced locally by this key. Since this is not the normal orientation, one test was run with a model rotated 90 deg to determine any adverse effects from testing at the unusual orientation. Figure 13 indicates that there are no adverse effects provided there is enough area to prevent a shear failure at the keyway.

No difference in cost figures between the rudder nut and the tapered key types of assemblies can be presented except on the model scale. It took about twice as long to fabricate and assemble the key end as it did the nut end although the material costs are about the same for the two assemblies.

When the final test was run, it was decided to investigate the effects of gross overload and test to failure. It was hoped to learn something about the failure mechanism of a joint of this nature. As the test progressed, however, it became apparent that the model was ductile enough that it would just bend excessively. At a load of 119,000 lb, which corresponds to a moment of 1,300,000 in-lb, the deflection was so great that the joint was considered operationally useless. Examination of the model after the load was removed showed the model to be badly deformed (Figures 16 and 17). The disassembled model revealed that the machined fit was badly distorted (Figure 16) and the tapered key was bent (Figure 18). No apparent thread damage was observed at the nut end.

The flexural stresses obtained during these tests are shown in Figure 19 for a moment of 900,000 in-lb. Although this moment was only two-thirds of the maximum applied moment, it was chosen since it was about the highest load at which most gages were still operative. It is noted that the data of Figure 19 do not fit the curve of Figure 14 due to yielding. However, by using a plot of head deflection versus load (Figure 20), it is possible t[illegible] t some idea of the onset of yielding in the model. If the flexural stresses are plotted for [illegible]oment of 520,000 in-lb, the onset of yield as shown by the head deflection (Figure 20), the results are compatible with those of Figure 14. To provide additional data, the results obtained in this way are plotted in Figure 14. The slight discrepancy is attributed to experimental scatter and nonsymmetry of loading.

SUMMARY AND CONCLUSIONS

Based on the data from these tests the following conclusions are drawn:

1. It is possible to predict the flexural stresses, within design accuracy, for a socketed connection loaded in pure bending if the stock penetrates the yoke at least one stock diameter.

2. It is impractical to predict the stresses on the face of the yoke. However, within the range of parameters considered in these tests, the face stresses are not large enough to cause concern.

3. Relief does not significantly affect the bending stresses in the yoke.

4. The bending stresses are about the same whether the rudder nut or the tapered key is used.

5. The 1 1/4-diameter model began to yield at a moment of 520,000 in-lb, but it was able to sustain a moment of 1,300,000 in lb without breaking. However, the bearing surfaces were badly deformed and a permanent deflection of approximately 1/2 in. was observed.

6. On the model scale, the cost of fabricating the nut was about half that of the key. However, this may be entirely different for full scale.

RECOMMENDATIONS

If a considerable amount of design work outside the scope of this program is planned, it is recommended that additional experimental work be done. This should be designed to extend the curves of Figure 14 particularly in terms of yoke thickness and taper ratio.

ACKNOWLEDGMENTS

The author wishes to express his thanks to all the members of the Model Basin staff who assisted with the tests. In particular, the author wishes to thank Mr. John Frazier, Industrial Department, for his assistance and helpful suggestions made during the model construction phase and to Mr. Robert Abrams who ran a number of the tests and reduced a great deal of the data.

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1. Joints--Design
2. Joints--Stresses--Measurement
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II. S-F013 03 01, Task 1978

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